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Experimental Investigation on Heat Transfer Enhancement in Laminar Flow in Circular Tube Equipped with Different Inserts

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***Abstract:*-An experimental investigation of heat transfer and friction factor of a smooth tube fitted with full length twisted tape inserts for laminar flow have been studied under uniform wall heat flux condition. The experiments has been carried out to study the tape fin effect by using full length tape inserts of different materials namely Aluminum, Stainless steel and insulated tape. The tapes have twist ratios from 5.2 to 3.4. It is found that, for the flow in smooth tubes, full length twisted tapes yield improvement in average Nusselt number, for Reynolds number range of 200 to 2000.For Aluminum tapes, the maximum improvement in Nusselt number range from 50% to 100%; for Stainless steel tapes, maximum improvement in Nusselt number range from 40% to 94% and for insulated tapes, maximum improvement in Nusselt number range from 40% to 67%. The isothermal friction factor for the flow with the twisted tape inserts are 340% to 750 % higher as compared with those of smooth tube flow, in the given range of twist ratios.**

***Index terms:*-Heat transfer enhancement, Twisted tape, Laminar flow, Twist ratio, Flow friction factor**

I. INTRODUCTION

Efficient utilization, conversion and recovery of heat are the predominant engineering problems of the process industry. The subject of enhanced heat transfer has developed to the stage that it was of serious interest for heat exchanger design. Heat transfer enhancement (HTE) techniques can be divided into two categories-passive and active. In passive HTE an object, which does not use any external energy, such as an insert, has the duty of increasing the heat transfer rate. There are three different approaches to the enhancement of tube-side convective heat transfer, namely, inserted devices, internal fins and integral roughness. Insert devices involve various geometric forms that are inserted in a smooth circular tube. To date large numbers of attempts have been made to reduce the size and costs of the heat exchangers. Despite the high-pressure drop the twisted tape insert generates considerable increase in heat transfer rate by formation of a swirling flow and increasing the turbulence intensity close to the tube wall.

A. Need for Augmentation of Laminar Flow Heat Transfer

In the laminar flow, heat transfer takes place mainly by conduction and molecular diffusion as there is no cross mixing of the fluid. Natural convection currents were also present. But the thermal conductivities of the fluids were low, with the exception of liquid metals. Therefore the heat transfer coefficients in laminar flow were generally low. So, for a given heat transfer rate, larger heat transfer whereas will have to be provided as compared with turbulent flow heat transfer situations. Hence an augmentative scheme is necessary in some cases to meet the size limitations imposed upon and for efficient operation.

II. LITERATURE SURVEY

Hong and Bergles [3] correlated heat transfer and pressure drop data for twisted tape inserts for uniform wall temperature conditions using water flow having Prandtl number in between 3 to 7 and ethylene glycol having Prandtl number ranging from 84 to 192. The authors found that Nu is a function of twist ratio, Re and Pr and it was 9 times that of empty tube.

Manglik and Bergles [4] experimented with twisted tapes for heating and cooling of ethylene glycol and reported augmentation of 300% heat transfer more than smooth tube values. Eiamsa-ard and Promvongue [5] reported the enhancement of heat transfer in a tube with regularly-spaced and full-length helical tape swirl generators, and concluded that the full-length helical tape with rod provide the highest heat transfer rate about 10% better than that without rod. Siva Rama Krishna, Govardhan Pathipaka, P. Sivashanmugam [6] investigates heat transfer characteristics of circular tube fitted with straight full twist insert and concluded that the heat transfer coefficient increases with Reynolds number and decreasing spacer distance with maximum being 50 mm. spacer distance for both the type of twist inserts. Saha and Dhal [8] investigated heat transfer enhancement in laminar flow using water as fluid with Prandtl number ranging from 205 to 518, having different configurations of twisted tapes. They found that friction factor and Nusselt number increases with insertion of any passive augmentation technique. Shivashnmgam and Suresh [9] carried out studies with twist ratios and spacer length 100, 200, 300, and 400 mm. For twist ratio 4.89 to 1.95, increase in Nusselt number was in the range 30% to 40% for Reynolds number in the range 200 to 2200 for full length helical twist and spacer length, whereas the decrease in friction factor was in the range 40% to 45%. Bhardwaj et al. [10] experimentally determined pressure drop and heat transfer characteristics for flow of water in a 75 start spirally grooved tube with twisted tape inserts. The conclusions were for constant pumping power smooth tube showed that the spirally grooved tube without tape yields

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maximum heat transfer coefficient of 400% in the laminar range and 140% in the turbulent range. Rahimi et al. [11] conducted experiment for study of heat transfer and friction characteristics in a tube equipped with modified twisted tape inserts and 31% increase in Nusselt number. The higher turbulence intensity of the fluid was expressed as the main reason for the experimental observations. Patil A G [12] investigated experimentally Heat transfer and flow friction in swirl flow inside 25 mm diameter tube. Reduced width twisted tapes of width ranging from 11.0- 23.8 mm, which are lower than tube inside are used. Reduced width twisted tape inserts give 18% to 56% lower isothermal friction factors than the full width tapes. Under uniform wall temperature Nussselt number decreases by 5% and 25% for tape widths of 19.7 and 11.0 mm respectively. Date [1] solved numerically the fundamental equations of continuity, momentum balance and energy balance for the fully developed laminar flow heat transfer in a tube containing twisted tape. The results were available only in the graphical form and predicted about 15% increase in heat transfer at Prandtl number of 150 and more.

III. EXPERIMENTAL WORK

A. Experimental Set-up

Fig.1 shows the experimental set-up used for the study was explained in this section. It consists of calming section, test section, Rota meters, and inlet tank with the capacity $0.3m^3$ for supplying water, outlet riser section, and water collection tank capacity $0.3m^3$ for receiving water from test section. Calming section with the dimension 3200 mm long, 15.8mm ID, 19.05mm OD made of brass tube was used to eliminate the entrance effect. The test section was of brass tube with the dimensions of 1200 mm long with 15.8mm ID, 19.05mm OD. An L-shaped connector was connected at the end of test section to mount riser section. Other side was connected to calming section through flange. The outside surface of the tube was brazed with Copper–Alumel, k type thermocouples which were used to measure the surface temperature of pipe. 12 thermocouples were mounted on surface of test section at regular distance to measure surface temperature. Two thermocouples, one at inlet and other at outlet were inserted inside tube to measure water inlet and outlet temperature respectively. The test section tube was wound with bare nichrome wire with resistance of approx. 2 &!/m was used as heating element. Mica sheet which acts as electrical insulator was wound on both sides of wire to avoid direct contact. Over the electrical winding, two layers of asbestos rope tape were wound. Over the asbestos tape winding approximately 50 mm thickness of glass wool was lined and over which, another two layers of asbestos rope tape was wound to minimize heat loss due to convection and radiation. The terminals of the Nichrome wire were attached to the Dimmerstat (Auto-transformer), by which heat flux can be varied by varying the voltage. A wattmeter was connected across the wire to measure input energy. Two calibrated glass rotameter having range 0.5 lpm to 5 lpm was used to measure flow over the full laminar ranges were attached to the calming section to measure the flow.

B. Experimental Procedure

While performing the experiment, electric supply was given to nichrome wire and waited till the steady state was reached. To identify weather steady state was reached; readings were taken after a particular time period till the two readings taken at successive time interval were same. Once the steady state was attained, inlet and outlet temperatures of the fluid were measured. Also pressure drop across the test section was also measured from manometer. To measure the input power wattmeter reading was also taken and supply voltage was noted. Readings were taken by keeping the input power constant and increasing the flow rate from 2gm/s to 20gm/s. Then readings were taken by increasing the input power from 120W to 640W and varying the mass flow rate. The readings were taken for plain tube without any insert and using Aluminum twisted tape of twist ratio 5.2 and 4.2, stainless steel twisted tape with twist ratio 5.2 and 4.2 and insulated tape having same twist ratio geometry as shown in Fig.2. In this manner, all the readings for different twist tapes for different flow rates and different power conditions were taken and tabulated.

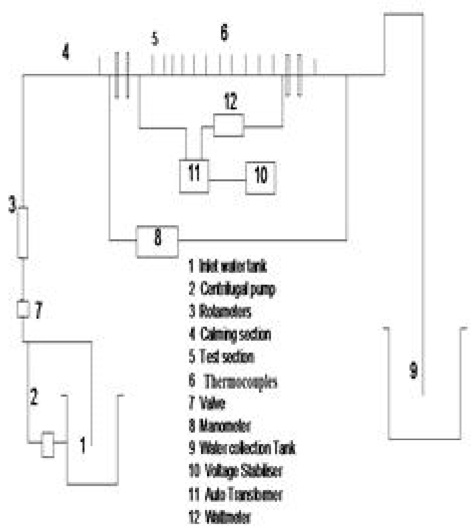
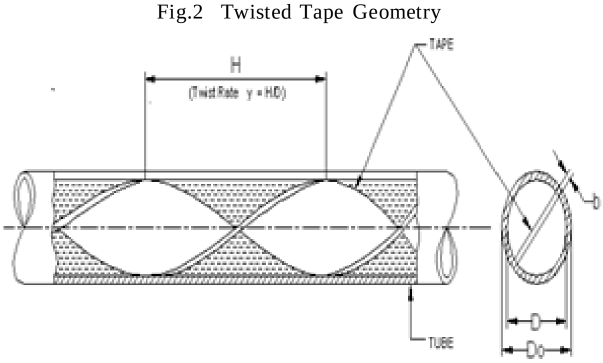


Fig.2 Twisted Tape Geometry



IV. THEORETICAL ANALYSIS

A. Effect of Twisted Tape Insert Technique

A semi analytical prediction for the Nusselt number for laminar flow heat transfer in a horizontal tube containing twisted tape was done and also a method to evaluate the mean isothermal friction factor was also studied. The total

heat transferred to fluid flowing in a tube containing a twisted tape may be assumed to be consist of following components

1) Heat transfer due to Centrifugal convection
2) Wall curvature effect i.e. Heat transfer due to forced and natural convection for flow over a curved surfaces
3) Heat transfer due to tape fin effect
4) Higher mean velocities
5) Roughness effect

$$\text{Hence } Nus = Nu_{cb} + Nu_{w} + Nu_{fin} \quad (1)$$

1. Centrifugal Convection

The centrifugal force field established by the helical flow increases the heat transfer when the tube wall was hotter than the fluid. Denser and colder fluid particles were flung out towards the tube wall displacing the lighter and hotter fluid particles present there. This effects greater mixing. The analysis of twisted tape was carried out by Manglik and Bergles [5] and gives a correlation

$$Nu_{cb} = 0.6466 \left[\frac{\pi\,\delta}{4yDi}\right]^{0.5} [Re]^{0.5}\, Pr^{1/3} \quad (2)$$

2. Wall Curvature Effect

Because of the twist experienced by the fluid flowing in a tube containing a twisted tape insert, the mean flow over the tube resembles that across a curved channel. Experimental and analytical investigations by A.E. Bergles [3] have shown that a turbulent mean velocity over a curved channel shows a remarkable difference over that on a flat channel even for a mild curvature, while no such difference exists in laminar flow.

An experimental correlation which agrees data of many investigations was given by A. E. Bergles [3]

$$Nu_{w} = [4.362 + [0.055 \{\frac{Gr\,Pr^{1.35}}{Pw^{0.25}}\}^{0.4}]^{2}\ ]^{0.5} \quad (3)$$

3. Tape Fin Effect

The tape fin effect refers to the heat transferred from the tube wall to the tape and from the tape to the fluid. If a good contact between the tape and the tube was ensured, a significant contribution from the fin effect can be expected. But in practical situations, to facilitate easy insertion and removal, a certain amount of clearance will be required which tends to reduce the fin effect. [3]

$$Nu_{fin} = 0.9\alpha \text{``} Nu_{o}\, C_{fin} \frac{[\cosh M - 1]}{\sinh M} \quad (4)$$

4. Higher Mean Velocity

The mean velocities in the tape generated swirl flow were higher due to,

i) The decreased effective area available for flow due to partial blocking of the tube by the tape of finite thickness.

ii) The helicoidal path to be followed by the fluid. Since the fluid has to traverse a longer path in helicoidal flow as compared with straight flow. The mean velocities will be higher than in straight flow for the same mass flow rate

5. Roughness Effect

Twisted tapes introduce additional surface into the flow field. The quality of the tape surface its roughness affects the frictional drag, the velocity distribution, the turbulence and the thickness of the fluid layer adjacent to the surface. Whereas the increased frictional losses in the tape generated swirl flow can be attributed mainly to tape surface friction, the turbulence promoted by roughness increases the heat transfer coefficients.

V. DATA REDUCTION

In order to study the heat transfer rate and friction factor characteristics of a uniform heat flux tube fitted with twisted tape, the heat transfer coefficient and pressure drop must be determined.

A. Heat Transfer

The steady state of the heat transfer rate was assumed to be equal to the heat loss from the test section which can be expressed as

$$Qa = Qconv \quad (5)$$

The heat transfer rate in the test section was calculated using

$$Qa = V^2/R = mCp(T_o - T_i) = U_o A_o (T_o - T_b) \quad (6)$$

Where

$$1/(U_o A_o) = 1/(h_i A_i) + \ln(Di/Do)\,(2\pi k_w L) \quad (7)$$

$$T_b = (To + Ti)/2 \quad (8)$$

The internal convective heat transfer coefficient hi was determined by combining Eqs. (7) and (8). The inlet temperature (Ti) and outer temperature (To) of water was measured at certain points with Cu–Al, k type thermocouples. The mean wall temperature can be expressed as

$$(Ti) = \text{``}\, Ti / 12 \quad (9)$$

Where Ti was the local inside temperature and evaluated at the outer wall surface of the inner tube. Twelve thermocouples were tapped on local wall of the tube and the thermocouples placed round the tube to measure the circumferential temperature variation, which was negligible. The average wall temperatures were calculated from 12 points, lined between the inlet and the exit of the test tube.

The mean Nusselt number, Nu was estimated as follows:

$$Nu = hD / k \quad (10)$$

The Reynolds number was given by

$$Re = VD/v \quad (11)$$

B. Pressure drop calculation

The pressure drop (ΔP) was determined from the differences in the level of fluid in U-tube manometer while fully developed friction factor was calculated using the following equation:

$$f = (Di/L)\left(\frac{\Delta P}{2\,\rho\, u_m^{2}}\right) \quad (12)$$

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All the fluid thermo physical properties determined at the average of the inlet and outlet bulk temperatures, T_b.
The uncertainties associated with the experimental data were calculated on the basis 95% confidence level. The measurement uncertainties used in the method were as follows: bulk fluid temperature and wall temperatures ± 0.1ºC, fluid flow rate ± 2%, and fluid properties ± 2%. The uncertainty calculation showed that maximum of ± 6%, ± 5%, and ± 8% for Reynolds number, friction factor, and Nusselt number, respectively.

VI. RESULTS AND DISCUSSION

Fig. 3 depicts variation of Nusselt number with Reynolds number for different twisted tape of twist ratio with mass flow rate increasing

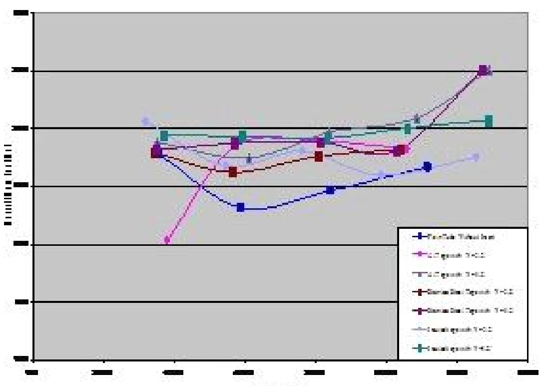

Fig.3 Nusselt Number Vs Reynolds Number

from 2gm/s to 20gm/s One can observe from fig.3 that the Nusselt number for full-length twisted tape increases with increasing mass flow rate and Reynolds number. This increase in intensity of heat transfer due to swirl flow generated in stainless steel full-length twisted tape is more than others. Maximum improvement in Nusselt no. using insulated tape of twist ratio 4.2 as 67%.It is interesting to note that in case of smooth tube, for a given heat flux as the Reynolds number increases, there is reduction in the Nusselt number. This can be attributed to the fact that, at a constant heat flux , larger flow rates tend to reduce the ratio of wall temperature to bulk mean temperature, thereby reducing Grashoff number and hence contribution from the natural convection.

The nature of distributions of data points shows the dependence of swirl flow, Nusselt number on heat flux also. The swirl flow Nusselt number is found to be the function of

- the twist ratio,
- Reynolds number and
- the heat flux.

Fig. 4 and Fig. 5 depicts in the Reynolds number range of 200 – 2000 the isothermal friction factor for flow with full length twisted tape insets are 340 % to 750% higher as compared with those of smooth tube flow in the range of twist ratios 5.2 to 4.2. The isothermal friction factor is the function of twist ratio and Reynolds number. At higher Reynolds number beyond laminar flow region, the friction factor increases due to increase in the momentum exchange of the molecules. The isothermal friction factor obtained in this experimental work

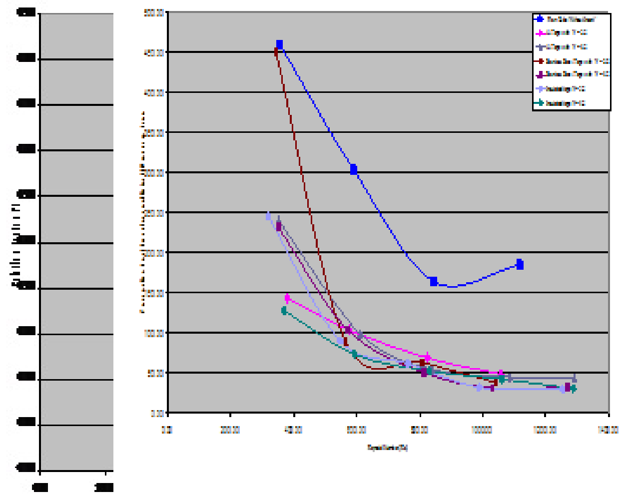

Fig.4 Friction factor (F1) Vs Reynolds Number (Re)

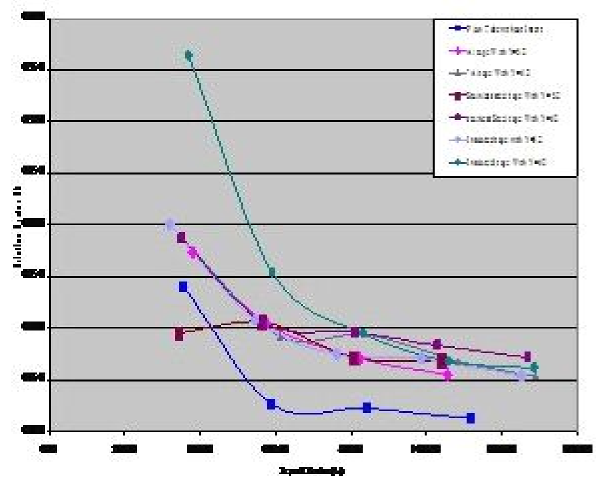

Fig.5 Friction factor (F2) Vs Reynolds Number (Re)

for flow with full length tape inserts are in excellent agreement with predicted values of Date[1]. Fig.6 shows comparison of effectiveness of tapes with smooth tube flow. The tapes were found to be effective over the range of Reynolds number from 200 to 2000. The contribution from the fin effect was marginal.

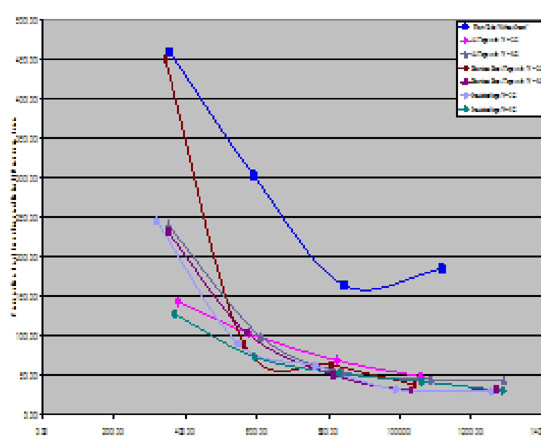

Fig.6 Convective heat transfer coefficient/ Pressure drop Vs Reynolds Number (Re)

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VII. CONCLUSION

Compared with heat transfer performance for flow in smooth tubes, full length twisted tape yield improvements in average Nu, for Re range of 200-2000 as follows,

1) In the range of given twist ratios, the aluminum tapes have shown a maximum improvement of 90% in the Nusselt number for y = 4.2 for Re = 2000
2) Aluminum tapes (Cfin = 8.15); between the twist ratios of 5.2 and 4.2, maximum improvements in Nu ranges from 50% to 100%.
3) S.S. tapes (Cfin = 0.61); between twist ratios of 5.2 and 4.2, the maximum improvement in Nu ranges from 40% to 94%.
4) Insulated tapes (Cfin = 0); between twist ratios of 5.2 and 4.2, the maximum improvement in Nu ranges from 40% to 67%.

VIII. FUTURE SCOPE

By using CFD techniques experimental result can be validated and used to adopt more accurate and precise Heat augmentation Techniques.

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NOMENCLATURE

A	heat transfer surface area, m^2
Cp	specific heat of fluid, J kg^{-1} K^{-1}
D_i	inside diameter of test tube, m
f	friction factor = $(Di/L)\ (\Delta P)/2\ \rho\ u_m^{\ 2}$
h	heat transfer coefficient, Wm^{-2} K^{-1}
k	thermal conductivity of fluid, W m^{-1} K^{-1}
L	length of test section, m
m	mass flow rate, kg s^{-1}
V	volume flow rate, m^3 s^{-1}
ΔP	pressure drop across test tube, N/mm^2
y	twist ratio
Q	heat transfer rate, W
T	temperature, °C
t	thickness of test tube, m
Re	Reynolds number = *VD/v*
H	pitch length of the twisted tape m
U	average axial flow velocity, m s^{-1}
R	resistance, &!
Nu	average Nusselt number = *hD/k*
Pr	Prandtl number = *μCp/k*
Gr	Grashoff number
Pw	wall parameter
u_m	bulk average fluid velocity (ms^{-1})
Cfin	fin parameter Cfin = $k_m\ \delta\ /\ k_f\ D_i$
ID	inner diameter
OD	outer diameter

Greek symbols

ρ	fluid density, kg m^{-3}
δ	tape thickness, m
μ	fluid dynamic viscosity, kg s^{-1} m^{-1}
v	kinetic viscosity, $m^2 s^{-1}$

Subscripts

b	bulk
conv	convection
i	inlet
o	outlet
p	plain tube
s	swirl generator
w	wall
Nu_{cb}	average Nusselt number by centrifugal convection
Nu_w	average Nusselt number by wall curvature effect
Nu_{fin}	average Nusselt number by tape fin effect

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